59th ILMENAU SCIENTIFIC COLLOQUIUM
Technische Universität Ilmenau, 11 – 15 September 2017
URN: urn:nbn:de:gbv:ilm1-2017iwk-090:9

ON THE INFLUENCE OF THE FLEXURE HINGE ORIENTATION IN PLANAR COMPLIANT MECHANISMS FOR ULTRA-PRECISION APPLICATIONS

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ABSTRACT

This paper presents the investigation of specific geometric parameters of planar compliant mechanisms with flexure hinges. The main focus is on the geometrical orientation of the flexure hinges and the form of the link connections between them. The synthesis of the compliant mechanism using a selected rigid-body model by replacing the hinges and the design process of the mechanism are shown. Therefore different parameters are selected and FEM simulations are done to investigate the motion behaviour. After the simulation process a special test bench is designed to measure the path of motion of prototypes and to verify the simulation results.

1. INTRODUCTION

Due to their advantages compared to conventional mechanisms, compliant mechanisms with prismatic flexure hinges are becoming increasingly important in precision engineering and micro technology. The presented approach for the synthesis of a compliant linkage mechanism is based on replacing the joints of the ideal rigid-body model by flexure hinges and to connect them by links with much higher stiffness. The purpose of the synthesis is to design a mechanism which has the advantages of a compliant mechanism while moving simultaneously on nearly the same path as its rigid-body model. Additionally there is a special attention to the reproducibility of the kinematic behaviour.

Besides other parameters, the kinematic behaviour of a compliant mechanism is affected by the geometric design of the flexure hinge contours. By using optimized flexure hinges with polynomial contours, the kinematic properties of compliant mechanisms can be improved. As a part of the synthesis, further geometric parameters which affect the design of the compliant mechanism are determined. One of these parameters which show a significant influence on the path of motion is the orientation of the flexure hinges.

The subject of this paper is to investigate the influence of the geometric parameters in addition to the hinge contour on the motion range, the stiffness and especially the path accuracy of a compliant mechanism by means of FEM simulations. Therefore, the parameters which affect the kinematic behaviour of compliant mechanisms are defined and classified. These parameters are analysed by means of a FEM simulation as a part of a systematic design of experiments using the example of a parallel-crank mechanism. To verify the simulation models, two prototypes with different orientations of the compliant parallel-crank mechanism are manufactured and tested by means of a special designed test bench. The results of measurement for the kinematic parameters are compared to those from analytic and numeric models. This way, the influences of the geometrical parameters, especially the orientation, of flexure hinges in a compliant linkage mechanism are shown.

2. SYNTHESIS OF THE COMPLIANT MECHANISM

The method for the synthesis of presented compliant mechanism is the replacement of the rigid-body model. It is a common approach for synthesis. Based on a selected rigid-body model the ideal joints are replaced by flexure hinges and connected by links with a significant higher stiffness.

2.1. Rigid-body model

The rigid-body model describes an ideal path of motion which is the reference for the following qualitative studies in addition to the exact straight line. The path of motion corresponds to an analytically predictable function, which results from the geometric parameters of the mechanism. The function can be characterized by the displacement in x and y direction of one particular point on the coupler of the mechanism. For the chosen example of the parallel crank, these parameters are the coupler length c and the crank length a. The point which is analysed during motion is the point K on the coupler. This point is called coupler point, see Figure 1. The displacement of this point is split in the two components v_x^* and v_y^* in dependence of the input angle φ^*.

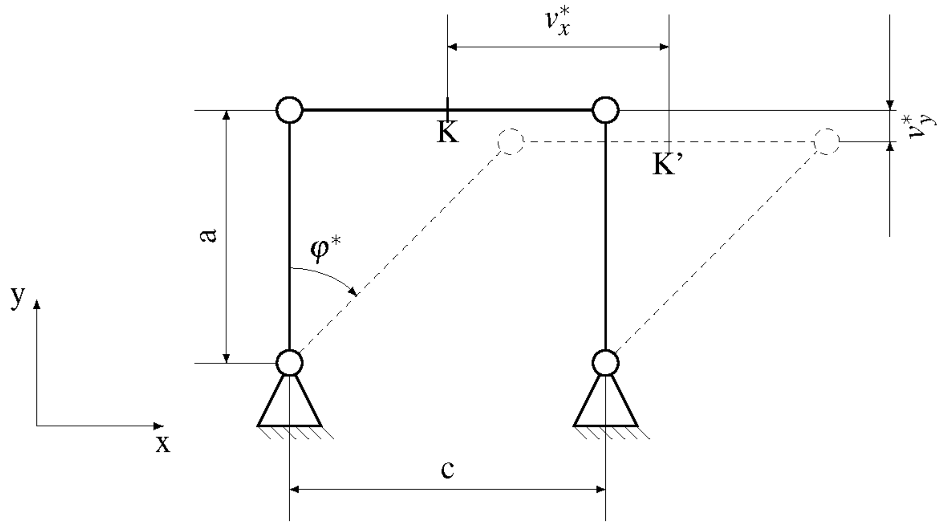

Figure 1 Rigid-body model of the parallel crank for the syntheses of a compliant mechanism with the coupler length c, the crank lenght a and the coupler point K [1]

In this ideal model the coupler point K moves, like every coupler point, on an ideal circular path of motion. This path is depending on the geometrical parameters a and c of the initial model. There is no bearing clearance or motion of rotation axes. The parameters for presented calculation, simulation and measurement are based on previous investigations [1]. In the shown case the crank length is $a = 80$ mm and the coupler length is $c = 100$ mm.

2.2. Compliant mechanism

The design process after the selection of a rigid-body model includes a lot of different parameters which show considerable influences on the motion behaviour of the system. In a first step the ideal revolute joints replaced by flexure hinges. Therefore the rotation axis of each flexure hinge is placed in the same position like the ideal joints. Also the geometrical form of the hinges must be selected. This includes all the parameters which are shown in Figure 2.

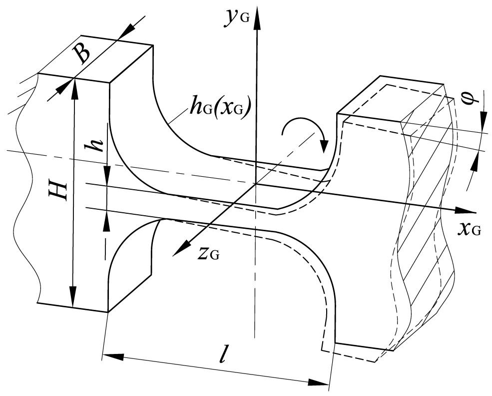

Figure 2 Single flexure hinge with its geometrical parameters: H hinge height, B hinge width, l hinge length and $h_G(x_G)$ function for the hinge contour [2]

3. GEOMETRICAL PARAMETERS IN THE DESIGN PROCESS OF A COMPLIANT MECHANISM

In the design process of the compliant mechanism there are a lot of geometrical parameters which could have influence on different output parameters of the system (e.g. path of motion, design space, load force). First, parameters were reflected in a previous investigation [2, 3]. This investigations were especially about the kinematic behaviour of a single flexure hinge and first compliant mechanisms without a view to the orientation of the hinges or the different possibilities for the geometry of the links between the hinges.

One of the results is that polynomial functions for the hinge contour have benefits with regard to the precision of path of motion, the stiffness of the mechanism and thus to the maximum stroke of it. With the help of the order of the polynomial function it is possible to optimise the compliant mechanism with regard to the requirements of the application. In reason of that by the investigations of additionally geometrical parameters a polynomial order of $n = 6$ is used in all hinges because the stress in all hinges is nearly the same.

3.1. Orientation of flexure hinges in compliant mechanisms

The synthesis of the compliant mechanism by the replacement of a selected rigid-body model describes the position of the rotation axes of the flexure hinges but not the orientation of the hinge. In the design process an angle for each hinge needs to be defined, which could be between 0° and 360°. By common mechanisms this angle is 0°, that the hinge length axis is in alignment with the adjacent links. A shift in the angle leads to a change in the arrangement of the links. In Figure 3 this is shown on the example of the parallel crank mechanism for three different angles in the upper hinges.

a) b) c)

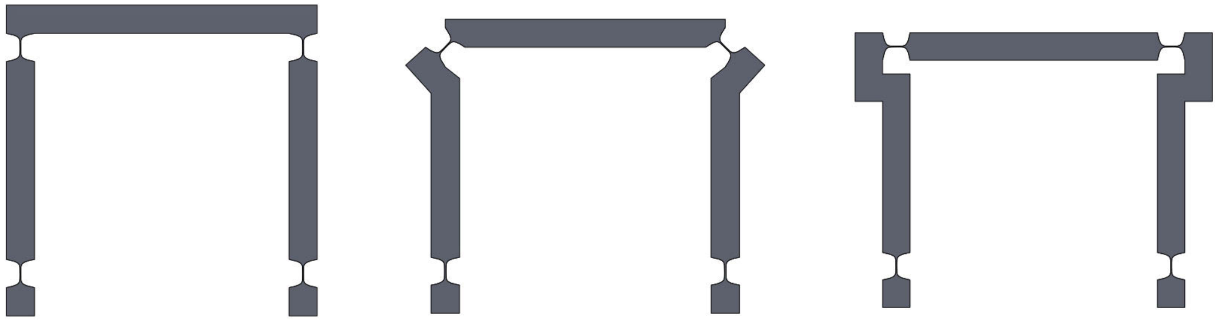

Figure 3 Compliant parallel crank mechanisms with different angles for the orientation of the flexure hinges which are not fixed to the frame a) 0° b) 45° and c) 90°

3.2. Design of the links between the flexure hinges

The design of the links between the hinges is like shown in the paragraph before at the one hand required to the change of the angle. It could also use as a separate parameter to manipulate the kinematic behaviour of the compliant system. The geometry of the links could be used to change the position of the connection to the links with the flexure hinge. This way the stress condition in the hinge can be changed. In Figure 4 two possibilities for the parallel crank mechanism with the same orientation of the hinges but with two different geometries of the coupler are shown.

a) b)

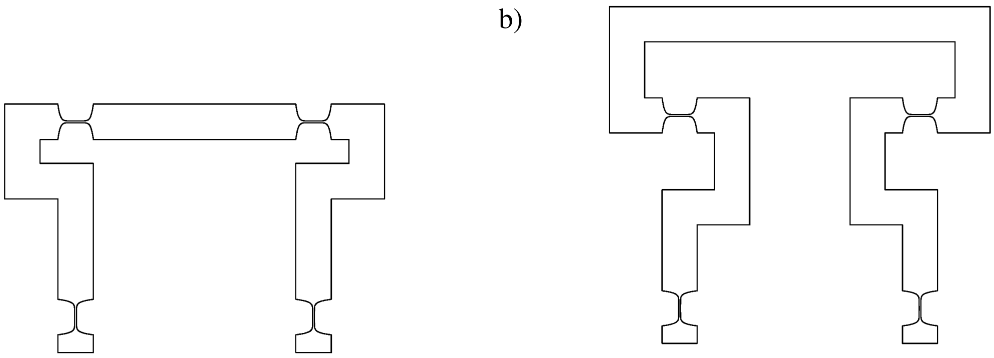

Figure 4 Different possibilities for the design of the links of a compliant parallel crank mechanism with the same oriantations of the hinge angles: a) coupler connects directly two hinges b) coupler connects indirectly two hinges

Geometrical adjustment of the links between the flexure hinges is an appropriated tool for the optimisation of compliant mechanisms.

With the improvement of the geometry of the links there is potential to vary the force application point, the point of the motion output, the relationship of the levers and the required design space.

4. FEM SIMULATION OF THE PATH OF MOTION

To investigate the path of motion of the different compliant parallel crank mechanisms simulations are done by FEM. After an intensive analysis of the simulation parameters like the conditions of the mesh and boundary conditions simulations are done. Several output parameters are analysed later on.

The simulation model was created using the software ANSYS Workbench 16.2 in consideration of large deflections. The deflection of the mechanism driven by a linear actuator is simulated by a force acting in x direction to the coupler. The coupler displacement in y direction and the rotation around the z axis are free. To obtain accurate simulation results comparable to the metrological investigation for the selected non upright position the following three aspects are considered in the FEM model too: The influence of gravity on the deformation in the z direction, a boundary condition to simulate the weight compensation implemented in the measurement setup by means of a guided ball and the weight of the measuring mirrors used for interferometric length measurement.

After specifying the input displacement the guiding properties of the coupler link can be determined. Thus, the position of the regarded coupler point K in the compliant mechanism results in comparison to the ideal straight line (straight line deviation v_y) and in comparison to the rigid-body model (lateral path deviation Δv_y). In addition, the angle of rotation δ of the coupler is calculated based on the coordinates of two points. The analysis of the maximum strain according to von Mises ε_V in the mechanism allows the calculation of the realizable stroke of the manufactured mechanisms according to the admissible strain for the chosen material. Also the increase of the input force while the motion is calculated by the FEM simulation. The results of the simulation are shown in paragraph 5.2 in comparison to the measurement results.

5. MEASUREMENT OF PROTOTYPES FOR A COMPLIANT PARALLEL CRANK

For a verification of the simulation results it is necessary to manufacture prototypes and measure the path of motion with a confidence level as high as possible.

Two different prototypes are manufactured by wire EDM, see Figure 5. The material is a hard aluminium alloy (AW EN 7075).

a) b)

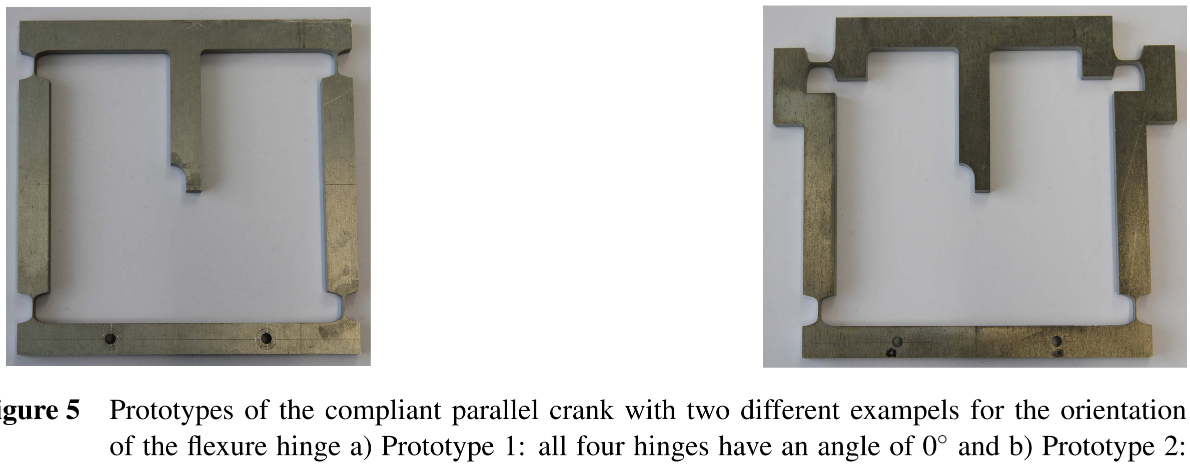

Figure 5 Prototypes of the compliant parallel crank with two different exampels for the orientation of the flexure hinge a) Prototype 1: all four hinges have an angle of 0° and b) Prototype 2: hinges which are not fixed to the frame have an angle of 90° (the form of the links between the hinges is selected under geometric conditions)

5.1. Test bench

The metrological investigation of the path of motion of the prototypes is made with a constructed test bench. The investigated parameters are corresponding to those of the simulation, the displacement v_x, the displacement v_y and the coupler rotation δ.

The two displacements can be determined directly be means of length measurements. To measure the rotation, it is necessary to use two parallel linear measurements carried out at a known distance and from their difference to determine the angle. The length measurements are made with plane mirror interferometers, which have a resolution of 0.1 nm. For realizing the described three length measurements, a single-beam interferometer and orthogonal a two-beam interferometer are used, see Figure 6. For the measurement of the input force F_i an additionally force cell is included, which is not shown in the picture.

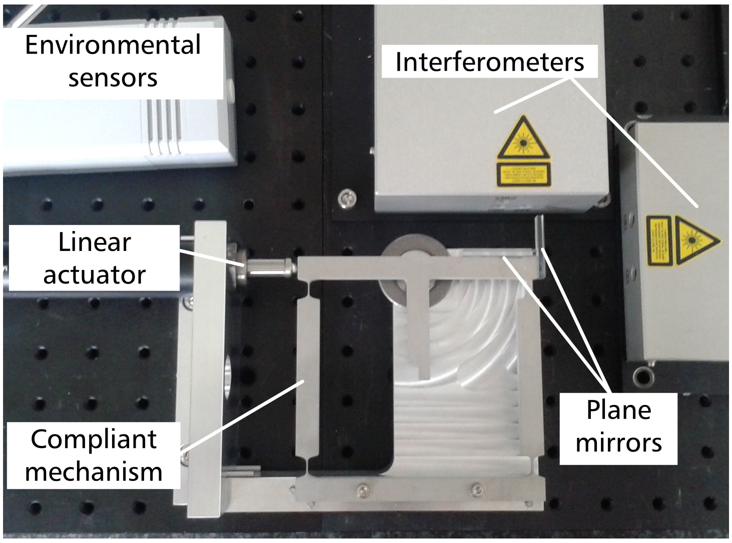

Figure 6 Test bench for measurement of the path of motion of the compliant parallel crank mechanism [1]

The initial driving motion is directly applied by a precision linear actuator at the coupler of the parallel crank, for all mechanisms. The resolution of the linear actuator is 0.1 µm. To compensate the dead load of the mechanism and the mirrors a rolling ball is used additionally, which is placed below the coupler and is moved along during deflection.

To ensure a statistically firm analysis, multiple measurements for each prototype have been made. The mean values are presented as functions in the following diagrams, while the indication of the confidence intervals is not shown due to a better clarity.

5.2. Results of measurement

Measurements on the presented test bench of the manufactured prototypes show a high conformity with the values from the FEM simulation. In Figures 7-10 the diagrams for the measured values are shown. In comparison to the analytic model the FEM simulations for two different compliant parallel cranks as well as the related measurement values are plotted.

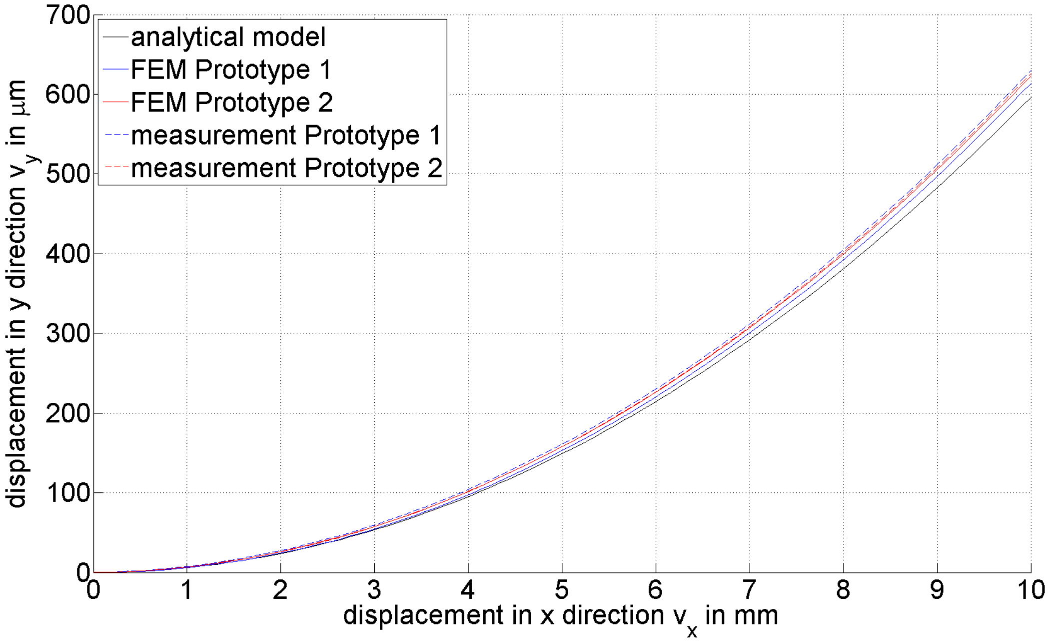

Figure 7 Results of measurement for the two manufactured prototypes in comparison to the analytical model and the simulation models by FEM for the path of motion

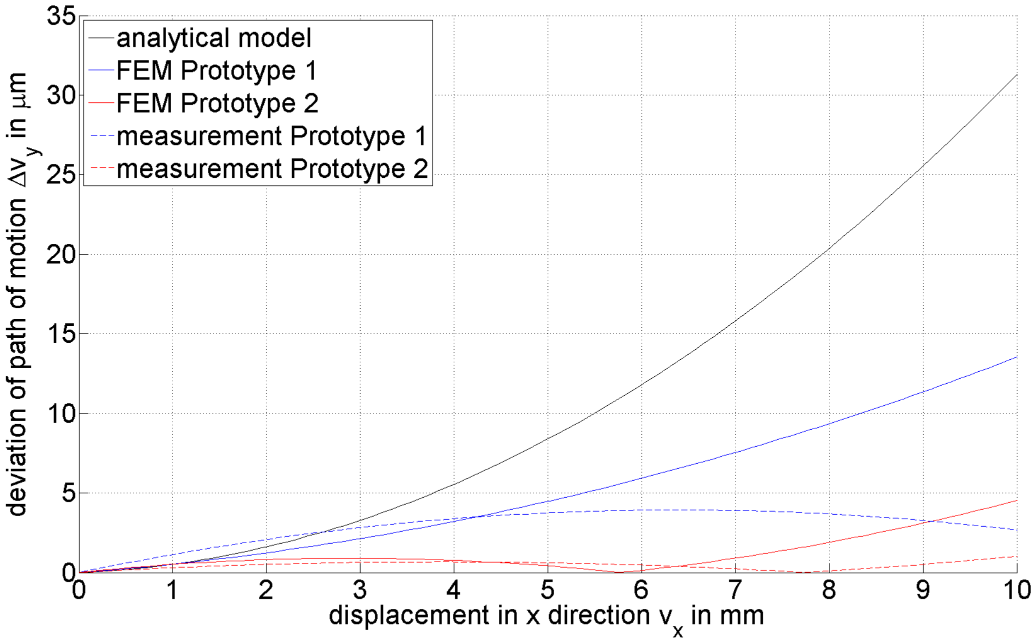

Figure 8 Results of measurement for the two manufactured prototypes in comparison to the analytical model and the simulation models by FEM for the deviation to path of motion for the rigid-body model

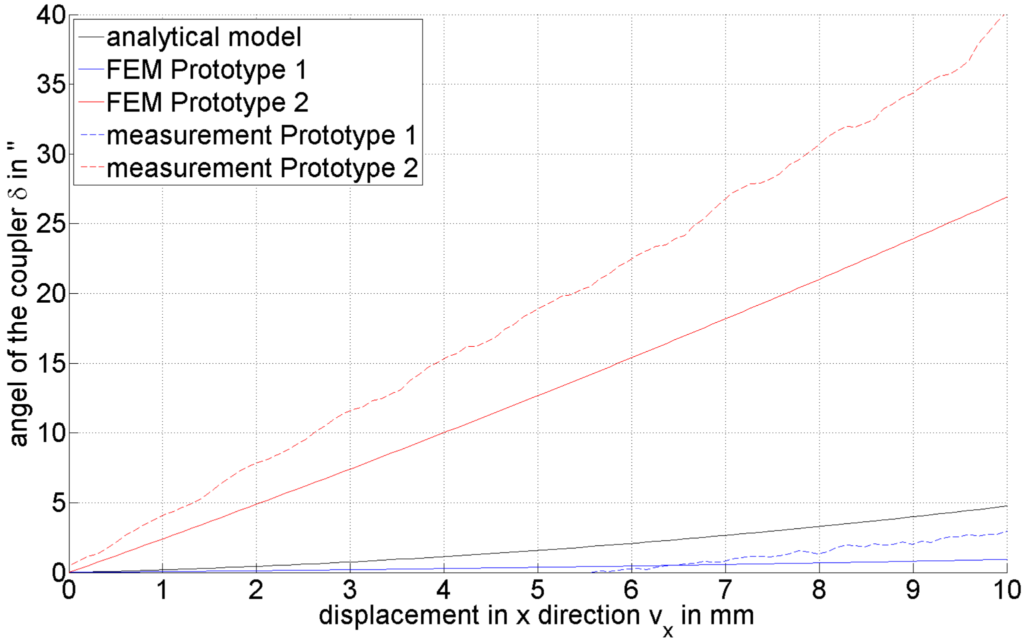

Figure 9 Results of measurement for the two manufactured prototypes in comparison to the analytical model and the simulation models by FEM for the rotation angle of the coupler

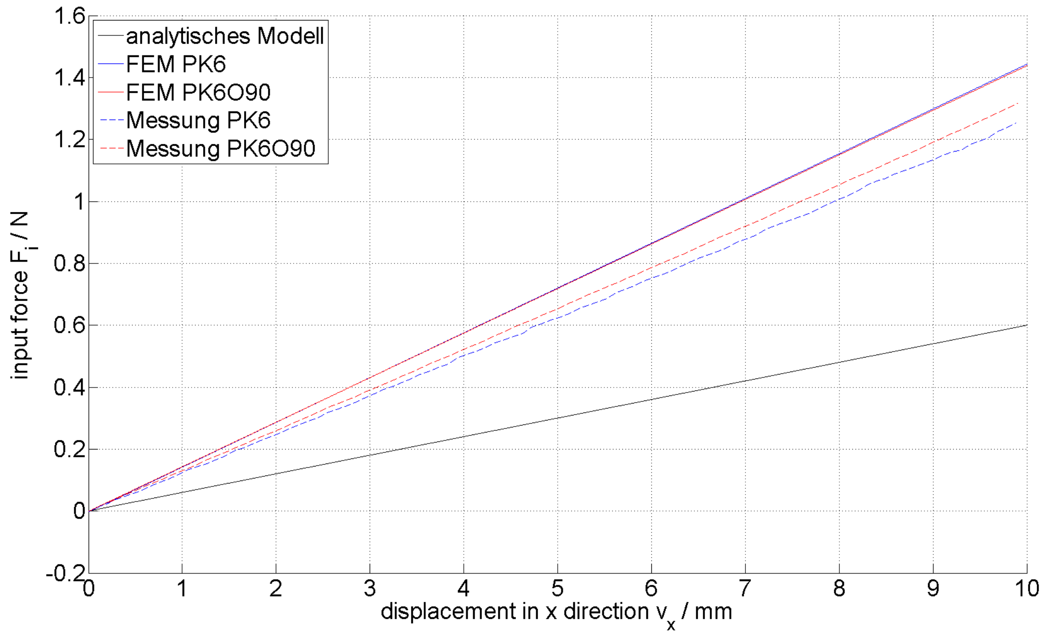

Figure 10 Results of measurement for the two manufactured prototypes in comparison to the analytical model and the simulation models by FEM for the input force of the linear actor

The very good conformity of the values and the implementation of the specifications for the path of motion proofs the selected way of synthesis and design for compliant mechanism.

6. CONCLUSION AND OUTLOOK

The investigations in the paper present a small selection of parameters for the design process of a compliant mechanism. It shows more design possibilities as only the selection of a rigid-body model or the variety of the flexure hinge contour. The geometrical parameters which are specifically shown in this paper, the orientation of the flexure hinges and the design of the links, are done in the most cases intuitive and elementary. But with the increase of the complexity of the system also the potential of variations for these parameters increase. On this way it is possible to create mechanisms with special motion behaviour for each ultra-precise application.

On the basis of simulation and measurement it was shown that a purposeful optimization of the presented parameters leads to an advantage in terms of reduced straight line deviations, less rotation angles or a smaller design space. The independence of the load force of the compliant mechanisms with different orientation of the flexure hinges show that there is negligible influence of the orientation to the stress in each hinge. So there is no influence on the optimisation of other parameters. For example the geometrical parameters of the flexure hinge. The influence of orientation of the hinge on the design of the links is obvious but in the most cases there are infinite possibilities.

In further investigations guidelines for the design of the parameters will be developed. This way the time for optimisation of compliant mechanisms can be severely reduced. Up to it is necessary to find complex optimisation algorithm or to do time-intensive variation studies for finding the best suitable solution to the application. Because of this in most application also in the field of ultra-high precision engineering the synthesis of the compliant mechanisms is shorten by using intuitive solutions or by using iterative manufacturing processes to get to the goal. Instead of the common intuitive design practice these opens a more strict systematic way allowing more precise and determined solutions.

ACKNOWLEDGMENTS

The development of this project is supported by the Deutsche Forschungsgemeinschaft (DFG) under Grant No. TH 845/5-1 and ZE 714/10-1.

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